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A comparison of advanced heat recovery power cycles in a combined cycle for large ships

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ABSTRACT

Strong motivation exists within the marine sector to reduce fuel expenses and to comply with ever stricter emission regulations. Heat recovery can address both of these issues. The ORC (organic Rankine cycle), the Kalina cycle and the steam Rankine cycle have received the majority of the focus in the literature. In the present work we compare these cycles in a combined cycle application with a large marine two-stroke diesel engine. We present an evaluation of the efficiency and the environmental impact, safety concerns and practical aspects of each of the cycles. A previously validated numerical engine model is combined with a turbocharger model and bottoming cycle models written in Matlab. Genetic algorithm optimisation results suggest that the Kalina cycle possess no significant advantages compared to the ORC or the steam cycle. While contributing to very high efficiencies, the organic working fluids possess high global warming potentials and hazard levels. It is concluded that the ORC has the greatest potential for increasing the fuel efficiency, and the combined cycle offers very high thermal efficiency. While being less efficient, the steam cycle has the advantages of being well proven, harmless to the environment as well as being less hazardous in comparison.

1. Introduction

The world seaborne trade has been growing in the last decades [1] and there is ever more motivation for reducing the pollution from large ships. Currently, international regulations on emissions of CO_2 (and energy efficiency) and oxides of sulphur and nitrogen (SO_x and NO_x) are changing towards stricter limits [1]. An important focus area in the efforts to reduce these emissions is WHR (waste heat recovery) systems. WHR systems are designed to recover engine waste heat and produce mechanical or electric power and if coupled with an engine shaft motor, a reduction of the main engine load is possible.

Being among the most common types of vessels in the current world fleet, the case in focus in this study is a feeder class container ship which has a typical capacity of 2500 TEU (twenty foot equivalent units) containers and a length of 200 m. Widely used in this class, and used as the case study here, is the MAN Diesel & Turbo 7L70 MC two-stroke low speed diesel engine with seven cylinders each with a bore of 70 cm, and a maximum continuous rating of about 20 MW.

Waste heat recovery systems are not yet standard in this class of vessels although solutions are currently available, namely either a steam Rankine plant or alternatively an exhaust gas power turbine. In the literature the most often mentioned alternatives to the steam cycle are the ORC (organic Rankine cycle) and the Kalina cycle. The ORC has been proposed for various applications of WHR including maritime ones for example by MAN Diesel & Turbo [2] as a WHR solution for smaller engines. The Kalina cycle has in the literature [3,4] been claimed to possess potential to achieve higher conversion efficiencies for WHR in general, compared to both ORC and steam Rankine cycles. Controversy exists however, and modelling efforts [5] have showed that the performance of ORC and Kalina may, at best, be similar for marine application. For this reason it is the goal of the present study to compare the three mentioned power cycles for the previously mentioned case study. Net power output of the cycles is the main parameter for comparison since it (when using a shaft motor) determines the resulting SFC (specific fuel consumption) and specific NO_x emissions of the combined cycle. In addition, important qualitative implications are considered in the comparison of the three different power cycles.

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Nomenclature

Acronyms

BOI	boiler
CAC	charge air cooler
CND	condenser
COM	compressor
ECO	economizer
EVA	evaporator
FWP	feed water pump
HP	high pressure
HPP	high pressure pump
IHX	internal heat exchanger
JWC	jacket water cooler
LP	low pressure
NO_x	nitrogen oxides
ORC	organic Rankine cycle
PTG	power turbine generator
REC	recuperator
SEP	separator
SFC	specific fuel consumption
SO_x	sulphur oxides
SUP	superheater
TC	turbocharger
TCC	turbocharger compressor
TCT	turbocharger turbine
TEU	twenty foot equivalent units
TUR	turbine
WHR	waste heat recovery

Though the concept of the mentioned combined cycle processes has been well described in the open literature e.g. by MAN Diesel & Turbo [2], modelling efforts for the design and optimisation are not. However, one well described example is Danov and Gupta [6,7], who presented a comprehensive mathematical model of a marine turbocharged diesel engine and a single pressure level steam Rankine WHR system including the associated auxiliary components. Validation of the model was presented at varied engine loads and speeds and the resulting fuel consumption was analysed.

This paper presents the result from similar modelling efforts with a model of a combined energy system, consisting of a large marine two-stroke low speed diesel engine, a turbocharger and a WHR system. Compared to previous work, this study also includes the estimation of NO_x emissions. This study further includes the comparison of three power cycles, a dual pressure steam cycle, a Kalina cycle and an ORC. In addition, the ORC optimisation method is more comprehensive, compared to previous work described in the literature, because it includes a combined optimisation of the process layout and the working fluid.

A description of the methodology is provided in Section 2. Section 3 presents the results from the modelling efforts and Section 4 discusses the results. Finally Section 5 compiles the main findings.

2. Methodology

Although the main engine model was presented in earlier work of the authors [8], it is briefly outlined in the following for the purpose of consistency. Then, the methodologies used for modelling the WHR power cycles are briefly outlined as to the degree suitable for the present format. Finally in this section, the applied optimisation algorithm is briefly described.

All the WHR models have been made using Matlab 2010b in combination with the NIST Refprop fluid property database [9] to provide the thermo-physical properties for the exhaust gas and working fluids. The WHR system turbines were modelled using a polytropic efficiency, in order to be able to evaluate the cycles at a wide range of pressures and to ensure a comparable level of technology for the expanders. Heat exchangers were modelled using energy balances that were divided into a suitable number of equal parts, in terms of heat transferred, in order to check for and avoid pinch point violations. The pumps were modelled using isentropic efficiencies and not polytropic ones in order to reduce computational time. No pressure or heat losses were accounted for in order to simplify the approach. This can be allowed since the analysis is aiming at providing generic results only.

2.1. Engine model

Various engine model methodologies are described in the literature, ranging from full 3D fluid dynamic models to black box models. The model described in the following is a zero-dimensional model, a type which has been shown [10–13] to provide good predictions of engine performance while being fast enough for energy systems optimisation. Previous work of the authors [8] presented the model and the validation of it. The model was validated on the ability to predict the performance of a marine low speed two-stroke engine in two different engine cases. Firstly, it was confirmed that the model could predict measured engine performance while being constrained by an extensive number of input parameters. Secondly, it was shown how the model responded to changes in engine tuning parameters with good accuracy. The validated parameters were those relevant for an electronically controlled engine, i.e. exhaust valve timing and variable injection timing. Also the effects of varying scavenging pressure were validated.

In the present study, the model estimates the brake power, waste heat flows, fuel consumption and NO_x emissions of the main engine. The evaluation is mainly done by means of an energy balance and a two-zone combustion model which is integrated over a single engine cycle. The combustion process is divided into intervals, and the product composition and the flame temperature are calculated in each interval. A double Wiebe function [14] is used to estimate heat release. Heat losses in the cylinder are estimated using the Woschni correlation [15]. The NO_x emissions are predicted using the extended Zeldovich mechanism [16].

The engine model utilises engine parameters from Goldsworthy [17] and the calibration was done as follows, to match measured performance data from the same source:

1. The end of injection timing was adjusted to obtain correct maximum cylinder pressure.
2. To obtain correct brake power output, the time of opening the exhaust valve was adjusted.
3. It was chosen to investigate a case of a mechanically controlled engine and therefore the best compromise for the exhaust valve opening time for both loads 75% and 100% was selected.

2.2. Steam Rankine cycle

Steam Rankine cycles are well described in the literature and proven in application. As inspiration for the steam Rankine cycle process layout in this study, was the plant currently proposed by

MAN Diesel & Turbo [2]. The process flow diagram is shown in Fig. 1.

Starting from (1) the working fluid (water) is at a low pressure. Running through an engine jacket water cooler heat exchanger (JWC) and then a charge air cooler heat exchanger (CAC), the water is preheated to the bubble point (3). The water stream is then split into two streams that run through a low pressure circuit (4) and a high pressure circuit (7) in the boiler. Hot exhaust gasses from the engine are running through the boiler from point (30) to (35). The low pressure stream is evaporated and superheated before entering the ST (steam turbine). The high pressure stream (7) is pumped to a high pressure, preheated, evaporated and superheated before entering the turbine. After the turbine (12), the stream is condensed.

The TC (turbocharger) is coupled to the process such that the exhaust gas from the engine is entering at (20) into a splitter which allows gas to be bypassed the TCT (TC turbine). The mass flow rate needed for the turbine to drive the compressor is lead through the turbine at (21) and the rest is bypassed and mixed with the cooler gas after the turbine. Charge air for the engine is compressed in the TCC (TC compressor) and is rejecting heat to the WHR process via the CAC at (40)–(42).

Validation of the steam cycle model was done using two different software, DNA [18] and the commercial tool Aspen Plus v. 7.2 [19]. The ability of DNA to model steam Rankine cycle power plants have been validated previously [20]. The results of the present model are the same as modelled results in DNA and Aspen Plus, see Table 6 in the Appendix.

2.3. Kalina cycle

The literature about the Kalina cycle is not comprehensive and the number of plants is modest at the present time, perhaps because the process was only invented in 1983 by Dr. Kalina [3] and because the technology is patented. The cycle is relatively complex and uses a mixture of ammonia and water as working fluid. Being a mixture, the working fluid evaporates and condenses non-isothermally and this potentially provides the advantage of a better match between non-isothermal heat sources and sinks [3]. The Kalina cycle has been proposed in many different configurations and in the present study, the model is made with a configuration similar to those found in work by Jonsson and Yan [21] and Bombarda et al. [5], which have been found suitable for marine application.

Fig. 2 illustrates the process flow diagram of the Kalina cycle and the TC. Starting from the FWP (feed water pump), the working fluid (1) is preheated in the JWC and further in a recuperator heat exchanger (REC 2). It then enters the boiler (BOI) where the exhaust gas from the engine (30) rejects heat in order to preheat, evaporate and superheat the working fluid. Power is extracted from the fluid in the turbine (TUR) where after the fluid give off heat in the recuperator (REC 1). The stream (6) is then mixed with another stream (16) before it is condensed. After the condenser, the stream (8) is pumped to an intermediate pressure before it is split into two streams (10) and (11). Stream (11) is heated via REC 1 and heated again via the CAC. The stream (13) then enters a separator which separates the stream into an ammonia lean liquid stream (14) and an ammonia rich vapour stream (17). Both of these streams give off heat in REC 2. The rich stream (18) is then mixed with stream (10) before it is condensed again.

As described by Marston [22], a key component in the cycle is the separator. To obtain maximum power, the stream running through the turbine (4), needs to be as ammonia rich as possible and at the same time the outlet pressure needs to be as low as possible. However, high ammonia concentrations require relatively high condensation pressures. Therefore, the separator supplies a stream of fluid with a relatively low concentration of ammonia (14), which is mixed with the turbine outlet stream (6). The separator also has to supply an ammonia rich stream (17) to be mixed with the stream that has been condensed (10), at a sufficient rate in order to restore the stream concentration which again will be running through the turbine.

For the separator to be able to deliver these concentrations and flow rates, the main concern is to ensure that the separator feed temperature is high enough and the feed pressure is low enough. The feed pressure is dictated by the need for condensation and thus depends also on the working fluid concentration and cooling water temperature and flow in condenser 2 (CND 2). Most important is the feed temperature which normally depends on the recuperation of energy from the turbine outlet stream. However, at low turbine outlet pressures there might not be enough heat to recuperate; hence there is a motivation for inserting the heat source of the CAC at this point.

Alternatively the CAC could be placed to preheat/evaporate the feed stream in points (2) and (3). However, by placing the CAC to heat the separator feed stream and then having a second recuperator (REC 2), it may be possible to recover most of the CAC heat for the feed stream. Another alternative would be to place the CAC as a source for reheating of the fluid after the turbine. However, these options were not explored in the present study due to the scope of the work.

Validation of the Kalina cycle model was done using the commercial tool Aspen Plus v. 7.2 [19] and a study by Leibowitz and Zervos [23]. The modelled results showed agreement within 1% of the numbers presented by Leibowitz and Zervos [23] and the results of the present model are the same as when using Aspen Plus, see Table 7 in the Appendix.

2.4. Organic Rankine cycle

The application of ORC plants is relatively new but rapidly rising in popularity. The main reasons for this seems to be the very high

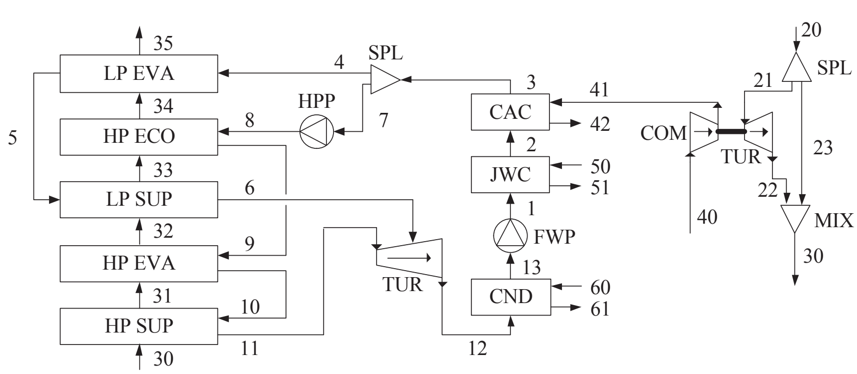

Fig. 1. Steam Rankine process flow diagram.

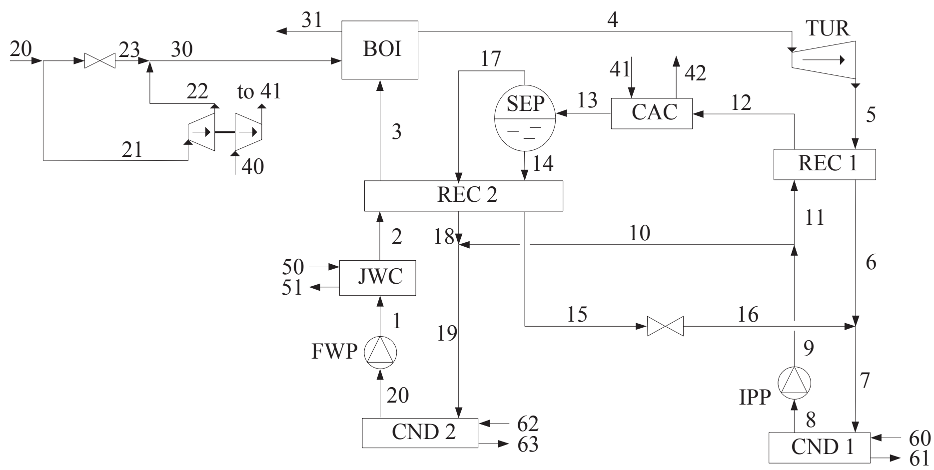

Fig. 2. Kalina cycle process flow diagram.

conversion efficiencies combined with a relative process simplicity, low cost, scalability and versatility. A key to the above properties of the ORC is the use of alternative working fluids, and thus the selection of optimum working fluid has received a lot of attention in the literature. A recent example is the work of Wang et al. [24] who investigated a number of refrigerant fluids for engine waste heat recovery.

There seems to be a general consensus in that there may to be no single fluid which can meet the requirements of a safe and optimally efficient working fluid. Usage of organic fluids raises issues with fire hazard, ecological hazards, corrosiveness, lubrication properties and cost. Therefore, the selection of working fluid is about finding the best compromise of properties and it is in that sense difficult to objectively state the optimum choice.

Fig. 3 illustrates the ORC process flow diagram. Working fluid enters the JWC at high pressure (1). Then it is heated further in a REC (recuperator) and in the CAC before entering the boiler. After the boiler, the fluid is expanded in a turbine before giving off heat in the recuperator (6). It is then condensed and pumped back to high pressure.

In the construction of the ORC process model used here, the optimisation algorithm was a strong factor. A total of 109 fluids from the Refprop fluid properties database have been tested using this model, and for that reason the model was made to adapt to the fluid characteristics. Most importantly in this regard is the fluid type; the wet type, which after expansion ends up as a mixture of liquid and gas such as water; the isentropic type, which has an isentropic saturation curve; and the dry type which after expansion is in a superheated state. A superheater is thus theoretically only needed for the wet type fluids.

Dry type fluids most often possess a great amount of heat after expansion and thus a recuperator is greatly beneficial to the overall cycle efficiency, whereas for wet and isentropic fluids, recuperation is most often not possible because the fluid is already at a saturated state.

Furthermore, due to the available heat from the jacket water and in many cases heat from recuperation, the heat available from the charge air may not be easily utilised. For these reasons an adaptable model was developed, such that the fluid properties, the different temperature levels and heat available could influence the process layout. The figure shows dashed lines around the CAC and the REC (recuperator) to illustrate that these components are optional for the process. Also the superheater is optional but in order not to prematurely discard any solutions of process and fluid, it was chosen to have the degree of superheating be an optimisation variable.

Many of the ORC working fluids are highly flammable and boiler leakages are therefore dangerous, risking explosions when highly flammable fluids come into contact with hot exhaust gasses. In order to avoid such a situation, an intermediate loop of heat transfer fluid (IHX) is transferring the heat to the boiler. A fluid designed for this purpose is a heat transfer fluid called DOWthermQ from the DOW chemical company and it was modelled as suggested by Pieronbon et al. [25].

Since the working fluids of the steam Rankine cycle and the Kalina cycle are non-flammable, this risk is not relevant and therefore no intermediate loop is used for the steam cycle and the Kalina.

Validation of the ORC model was done using the commercial tool Aspen Plus v. 7.2 [19] and the results of the present model are very similar as when using Aspen Plus for a range of tested working fluids, see Table 8 in the Appendix.

2.5. Power turbine generator

An alternative to the WHR systems described above is the PTG (power turbine generator), which is an exhaust gas powered gas turbine. The PTG is able to utilise the pressure component of the exhaust gas energy. The major part of the exhaust passes through the turbochargers, however, due to the excess turbocharger efficiency at high loads, some of the gas is bypassed the turbocharger and can be utilised by the PTG. The PTG is suggested as a cost-effective alternative for engine sizes similar to the presently studied [2]. It is in the present study assumed to have the same efficiency as the turbocharger turbine.

2.6. Optimisation

Building on the principles of natural selection, the Genetic Algorithm [26] is an optimisation algorithm which optimises multiple parameters for any given model. The parameters to be optimised are emulated as genes of individuals which are part of a population. The fittest individuals are combined, as in nature, to form subsequent generations of individuals. The algorithm uses a stochastic approach to form the first generation of individuals. In the presented work, the genes were the parameters for the WHR system, i.e. the boiler pressure, and in the case of ORC and Kalina, the fluid and solution concentration, respectively. For the steam plant, the mass flow rate fraction in the high pressure circuit was optimised and for the ORC, the superheater approach was optimised. The turbine outlet pressure was optimised for the Kalina cycle only.

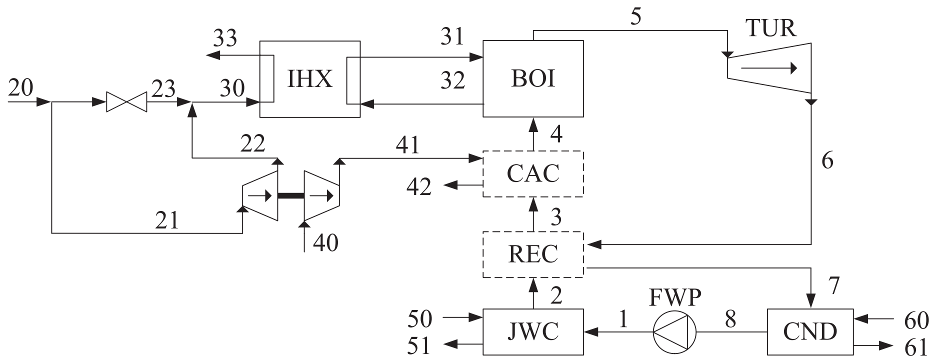

Fig. 3. Organic Rankine cycle process flow diagram.

2.7. Modelling parameters and conditions

Table 1 lists the design and operation parameters used for the WHR process models. The exhaust gas temperature at the boiler exit was limited to 160 °C for the prevention of sulphuric acid corrosion on heat exchanger surfaces. Since the TC compressor and the TC turbine operate at a limited pressure range and at the same conditions for all simulations, using an isentropic efficiency was assumed adequate. Efficiencies for the TC were calculated from the calibration case (see Section 2.1) at loads 75% and 100%.

It is noted that the minimum boiler pinch point temperature difference is a minimum that is allowed to vary and assume higher values during optimisation.

3. Results and analysis

First in this section, results from the main engine calibration and tuning are presented. Following, the combined cycle performances are compared. Finally, a qualitative evaluation of other relevant aspects concerning each of the cycles is presented.

3.1. Engine model

Table 2 presents the calculated outputs from both the calibration efforts and the engine tuning (designated WHR engine). Calibration data is seen in the parentheses for 75 and 100% load cases, and these were obtained at ISO ambient reference conditions, i.e. 25 °C and 1 bar pressure.

At 100% load, the model underestimates the power output, while it is overestimated at 75% load. Conversely, the SFC is overestimated at 100% load and underestimated at 75%. The deviations are within 1% accuracy while NO_x emissions are predicted with 5–10% accuracy.

Except for the jacket water heat, the overall energy balance of the engine seems also to be predicted accurately. In the work of Goldsworthy [17], the source of the calibration values, it is not made clear whether lubrication oil heat from the engine is included in the stated amount of heat from the engine. Should that be the case, the calculated results are with an accuracy of 5–10% of the reference values.

An effort was made to tune the main engine such that exhaust gas temperature would be suitable for WHR at a design point of 85% MCR. As it was chosen to explore the case of a mechanically controlled engine and therefore the exhaust valve timing was constrained to the value of the standard case. However, the effects of changes in injection timing, scavenging pressure, fuel and air mass flow rates and cylinder wall temperature were investigated. A targeted increase of the exhaust gas temperature of 50–65 °C was set as is in accordance with what is stated by MAN Diesel & Turbo [2]. The exhaust gas temperature after the TC for the standard tuning 85% load is seen to be 179 °C and WHR is assumed to be infeasible in this case. A combination of 10% lower charge pressure, 10% lower inlet air mass flow rate and an increase of 100 °C of the averaged cylinder wall temperature, was found to be a fuel effective way to gain an increase of 55 °C. It is seen in Table 2 how this tuning also causes reduced jacket water heat and temperature of the charge air.

3.2. Combined cycle

A comparison of the calculated performance for the combined cycle is shown in Table 3. The results suggest that the maximum obtainable net power production is highest for the ORC. The steam

Table 1
Design and operation parameters.

Minimum superheater approach, °C	20
Exhaust gas temperature after boiler, °C	160
Minimum boiler pinch point temperature difference, °C	10
Minimum turbine steam quality, %	85
Condenser working fluid outlet temperature, °C	40
WHR turbine polytropic efficiency, %	80
Power turbine isentropic efficiency, %	89
Pump isentropic efficiency, %	80
TC compressor isentropic efficiency, %	84
TC turbine isentropic efficiency, %	89
Charge air cooler pinch point temperature difference, °C	10
Jacket water cooler pinch point temperature difference, °C	5
Recuperator pinch point temperature difference, °C	10

Table 2
Engine model outputs.

Performance characteristics	Standard	Standard	Standard	WHR engine
Load, % maximum continuous rating	100	85	75	85
Power, MW	19.66 (19.81)	16.93	14.92 (14.86)	16.93
SFC, g/kWh	175.3 (174.0)	170.6	170.4 (171.1)	173.3
NO_x, g/kWh	14.5 (13.6)	15.8	16.3 (17.6)	17.3
Maximum pressure, bar	141.0 (141.0)	135.5	126.0 (126.0)	128.3
Exhaust temperature before TC, °C	344 (333)	302	281 (284)	344
Exhaust temperature after TC, °C	204	179	173	234
Charge air temperature after TC, °C	181	161	146	148
Fuel mass flow, kg/s	–	0.115	–	0.1165
Exhaust mass flow, kg/s	52.1	46.2	41.9	42.1
Charge air mass flow, kg/s	51.2	45.4	41.2	41.3
Jacket water heat, MW	2.28 (3.00)	2.24	2.18 (2.40)	2.16

Table 3
Combined cycle performance.

	Engine	Steam	ORC	Kalina	PTG
WHR power production, MW	–	0.863	1.16	0.825	0.453
Total power production, MW	16.94	17.8	18.1	17.76	17.39
SFC, g/kWh	170.6	164.9	162.2	165.3	168.8
NO_x, g/kWh	15.8	16.6	16.2	16.7	16.8
Combined efficiency, %	49.4	51.1	52.0	51.0	49.9

Rankine produces only about 75% of the power of the ORC and the Kalina process has a similar output. The power specific fuel consumption and NO_x emissions are reduced accordingly.

The results indicate that by using the ORC WHR system, the SFC can be reduced by 5% while NO_x increases slightly due to the generally higher in-cylinder temperature. The overall plant efficiency is 52.0% with the ORC compared to 49.4% without WHR. Table 3 also shows the potential output of a stand-alone power turbine generator. Being much less complex, the PTG produces a little more than half of the power produced by the Kalina cycle.

Table 4 presents the optimised parameters of each of the cycles. It is clear that the boiler pressure required to reach optimum power output is relatively high for the Kalina cycle compared to the others and this may influence the overall cost and safety precautions of the plant negatively.

It is notable that even though the ORC process optimisation algorithm was allowed to search for optimum also in the supercritical state domain, the optimum was found with a fluid just below the supercritical pressure. The optimal process layout of the ORC using R245ca as working fluid is having almost no superheating and thus this heat exchanger is not needed. R245ca is a HFC refrigerant fluid also known as penta-fluro-propane.

Fig. 4 illustrates the optimum boiler heat transfer characteristics for comparison. The working fluid temperatures entering the boiler are similar for steam and ORC processes while about 40° C lower for the Kalina process. This indicates that the Kalina cycle utilises the jacket water and charge air heat less efficiently. The ORC process uses the jacket water, recuperator and then charge air cooler heat to preheat the working fluid to 140 °C, just as high as the steam cycle. Note that for the ORC process the heat source is the heat transfer fluid (abbreviated DOW).

It was further investigated to use a non-flammable working fluid in the ORC and thus be able to simplify the process and remove the intermediate heat transfer circuit. The optimum net power obtainable for such a system was found to be 1060 kW and in the combined cycle 18.00 MW, with R236ea working fluid at the supercritical pressure of 68.3 bar. This is quite close to the optimum power of 18.10 MW found using R245ca in the ORC in Table 4. R236ea has a global warming potential of 1200 (CO_2-equivalent 100 years horizon) compared to 560 for the R245ca fluid [27], and thus represents an increase in environmental impact.

3.3. Qualitative comparison

Other aspects are drawn into the comparison of the three power cycles for a more complete analysis. In Table 5 each of the plant

Table 4
Combined cycle performance.

Steam		ORC		Kalina	
Boiler pressure high, bar	9.8	Fluid	R245ca	Boiler pressure, bar	86.6
Boiler pressure low, bar	3.4	Boiler pressure, bar	37	Turbine outlet pressure, bar	5.1
High pressure mass flow, %	65.7	Superheater approach, °C	52.2	Boiler ammonia concentration, %m.	76.4

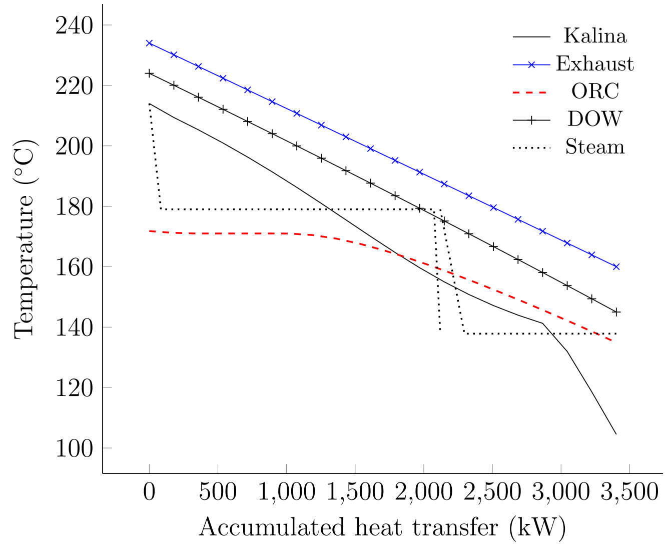

Fig. 4. Boiler heat transfer diagram for three power cycles alternatives.

options are given a minus, zero or a plus to indicate a relative qualitative disadvantage, a neutral evaluation or an advantage, respectively.

The plant size estimation is based on number of components and working fluid density. A high density results in comparatively small equipment sizes [28] and R245ca has about 5 times higher density compared to both water and ammonia-water mixtures.

Each of the aspects may be weighted subjectively according to particular needs and requirements. It does seem, however, that the Kalina cycle cannot present any advantages in this context. The ORC presents advantages of being compact in size and the highly efficient with a relatively simple process layout. The steam cycle, being the industry standard, seems to be a desirable choice due to its high efficiency, proven technology and very good environmental profile.

4. Discussion

With the applied methodology it seems possible to quantify the performance improvement of utilising either of the three power cycles as WHR for marine applications. The accuracy of the results does however rely on the reliability of the thermodynamic equations of state, residing in the Refprop database. Refprop have been used for the modelling of various kinds of systems and has been validated throughout the literature, one example is Colonna [29]. Furthermore to the accuracy of the results; since the models in this study were made without any pressure and heat losses, the results should be interpreted as ideal, and thus smaller net power outputs can be realised in an actual plant.

While the investigations in this study were made for a single main engine load point, the decisions regarding WHR plant process layout and working fluid may rely on a range of load points. A combined optimisation of the main engine with TC and WHR

Table 5
Comparative qualities.

	Steam	Kalina	ORC
Net power output	0	–	+
Known technology	+	–	0
Process complexity	0	–	+
Toxic working fluid	+	–	–
Hazardous working fluid	+	0	–
Environmental concerns	+	0	–
Plant size	0	0	+

versus a typical voyage load profile is therefore desired and subject for future work.

According to a MAN Diesel & Turbo report [2], the steam WHR system may produce about 10% extra power where in the present study an increase of only 5% was found. The combined cycle efficiency in the report is stated to be 55% where in the present study 52% efficiency was achieved. The following reasons could be contributing to this discrepancy: Firstly, in the MAN Diesel & Turbo report the WHR system includes both a Rankine cycle and a power turbine. Secondly, the fuel consumption in the present study was increased by 2.7 g/kWh while in the mentioned report this number is 10 g/kWh. This indicates that the tuning involves a richer air fuel mixture in the main engine, compared to the tuning done in the present work. Consequently, the additional fuel input would increase the exhaust gas temperatures and mass flow rates and cause a significantly higher WHR system power output. In the previously mentioned report, the exhaust gas temperature entering the WHR system is 285 °C compared to the 234 °C shown in Table 2. Lastly, a contribution may be that the WHR turbine efficiency of 80% used in the present work was assumed too low, and thus the respective power outputs may have been underestimated.

5. Conclusion

With the use of a previously derived marine two-stroke low speed diesel engine model, a simple turbocharger model and models of steam Rankine, Kalina and ORC WHR systems, the performance of the combined cycle was investigated. The engine model was tuned to produce a higher exhaust gas temperature for WHR system application. It was found that the tuning caused an increase in main engine fuel consumption, but the combined cycle fuel consumption was shown to be lower.

The results indicated that the ORC contributed with about 7% additional power and the Steam Rankine and Kalina cycles contributed with about 5% additional power. The application of a power turbine generator was found to be able to produce 2.5% additional power. In the best case, the specific fuel consumption was reduced from 170.6 to 162.2 g/kWh with the ORC WHR system though with increased NO_x emissions in exchange, due to the engine tuning.

When looking at other aspects than the mentioned, the Kalina cycle did not seem to offer any advantages. Instead, a number of drawbacks can be attributed, namely that the cycle is a relatively complex, unproven process utilising a toxic working fluid. The ORC plant offers the simplest plant layout with the highest efficiency. Drawbacks for the ORC are mainly related to the working fluid which is hazardous and environmentally damaging and perhaps it may also be a drawback that it is a relatively untested process in marine applications.

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Appendix

T, P, h, x, $\dot{m}$ and y_{NH_3} are temperature, pressure, specific enthalpy, vapour quality, mass flow rate and ammonia concentration (by mass).

Table 6
Steam model state points and process power.

	Matlab with Refprop					Aspen Plus				
	T (°C)	P (Bar)	h (kJ/kg)	x (−)	$\dot{m}$ (kg/s)	T (°C)	P (Bar)	h (kJ/kg)	x (−)	$\dot{m}$ (kg/s)
1	40.0	3.4	167.9	–	1.501	40.0	3.4	167.9	–	1.501
2	80.0	3.4	335.2	–	1.501	80.0	3.4	335.2	–	1.501
3	137.8	3.4	579.9	0.00	1.501	137.8	3.4	579.9	0.00	1.501
4	137.8	3.4	579.9	0.00	0.515	137.8	3.4	579.9	0.00	0.515
5	137.8	3.4	2730.6	1.00	0.515	138.2	3.4	2731.4	1.00	0.515
6	179.0	3.4	2820.1	–	0.515	179.4	3.4	2820.9	–	0.515
7	137.8	3.4	579.9	0.00	0.986	137.8	3.4	579.9	0.00	0.986
8	137.8	9.8	580.3	–	0.986	137.9	9.8	580.7	–	0.986
9	179.0	9.8	758.6	0.00	0.986	179.0	9.8	758.6	0.00	0.986
10	179.0	9.8	2776.3	1.00	0.986	179.0	9.8	2776.3	1.00	0.986
11	214.0	9.8	2862.6	–	0.986	214.0	9.8	2862.5	–	0.986
12	40.0	0.074	2272.3	0.87	1.501	40.0	0.074	2239.6	0.87	1.501
13	40.0	0.074	167.5	0.00	1.501	40.0	0.074	167.5	0.00	1.501
30	234.0	1.2	690.4	–	42.1	234.0	1.2	690.4	–	42.1
31	232.2	1.2	688.3	–	42.1	232.2	1.2	688.4	–	42.1
32	189.0	1.2	651.1	–	42.1	189.0	1.2	651.1	–	42.1
33	188.0	1.2	640.0	–	42.1	188.0	1.2	640.0	–	42.1
34	184.2	1.2	635.8	–	42.1	184.2	1.2	635.8	–	42.1
35	160.0	1.2	609.5	–	42.1	160.0	1.2	609.5	–	42.1
41	148.0	2.9	548.6	–	41.3	148.0	2.9	548.6	–	41.3
42	139.3	2.9	539.7	–	41.3	139.3	2.9	539.7	–	41.3
50	85.0	3.0	356.2	–	52.8	85.0	3.0	356.2	–	52.8
51	83.9	3.0	351.4	–	52.8	83.9	3.0	351.4	–	52.8
FWP power (kW)				0.629		0.629				
HPP power (kW)				0.850		0.850				
LP turbine power (kW)				250		250				
HP turbine power (kW)				614		614				
Cycle net power (kW)				862		862				

Table 7
Kalina model state points and process power.

	Matlab with Refprop						Aspen Plus					
	T (°C)	P (Bar)	h (kJ/kg)	x (−)	$\dot{m}$ (kg/s)	y_{NH_3} (−)	T (°C)	P (Bar)	h (kJ/kg)	x (−)	$\dot{m}$ (kg/s)	y_{NH3} (−)
1	41.8	86.6	297	–	2.33	0.764	41.8	86.6	297	–	2.33	0.764
2	80.0	86.6	485	–	2.33	0.764	80.0	86.6	485	–	2.33	0.764
3	104.5	86.6	613	–	2.33	0.764	104.5	86.6	613	–	2.33	0.764
4	214.0	86.6	2073	–	2.33	0.764	214.0	86.6	2073	–	2.33	0.764
5	95.7	5.1	1703	0.85	2.33	0.764	95.7	5.1	1703	0.86	2.33	0.764
6	56.0	5.1	1147	0.62	2.33	0.764	56.1	5.1	1148	0.63	2.33	0.764
7	76.8	5.1	720	0.30	5.20	0.493	76.8	5.1	721	0.31	5.20	0.493
8	40.0	5.1	99	0.00	5.20	0.493	40.0	5.1	99	0.00	5.20	0.493
9	40.1	11.6	100	–	5.20	0.493	40.1	11.6	100	–	5.20	0.493
10	40.1	11.6	100	–	0.73	0.493	40.1	11.6	100	–	0.73	0.493
11	40.1	11.6	100	–	4.47	0.493	40.1	11.6	100	–	4.46	0.493
12	77.5	11.6	390	0.09	4.47	0.493	77.5	11.6	390	0.09	4.46	0.493
13	114.5	11.6	958	0.36	4.47	0.493	114.5	11.6	958	0.37	4.46	0.493
14	114.5	11.6	410	–	2.87	0.273	114.5	11.6	410	–	2.87	0.273
15	106.8	11.6	374	–	2.87	0.273	106.8	11.6	374	–	2.87	0.273
16	89.0	5.1	374	0.05	2.87	0.273	89.0	5.1	374	0.05	2.87	0.273
17	114.5	11.6	1941	1.00	1.60	0.888	114.5	11.6	1941	1.00	1.60	0.888
18	106.8	11.6	1820	0.95	1.60	0.888	106.8	11.6	1820	0.95	1.60	0.888
19	88.4	11.6	1280	0.65	2.33	0.764	88.4	11.6	1279	0.65	2.33	0.764
20	40.0	11.6	284	0.00	2.33	0.764	40.0	11.6	283	0.00	2.33	0.764
30	234.0	1.2	690.4	–	42.1		234.0	1.2	690.4	–	42.1	
31	160.0	1.2	609.5	–	42.1		160.0	1.2	609.5	–	42.1	
41	148.0	2.9	548.6	–	41.3		148.0	2.9	548.6	–	41.3	
42	87.5	2.9	487.2	–	41.3		87.5	2.9	487.2	–	41.3	
50	85.0	3.0	356.2	–	52.8		85.0	3.0	356.2	–	52.8	
51	83.0	3.0	347.9	–	52.8		83.0	3.0	347.9	–	52.8	
FWP power (kW)						31.5						31.5
IPP power (kW)						5.2						5.2
Turbine power (kW)						863						863
Cycle net power (kW)						826						826

Table 8
ORC model state points and process power.

	Matlab with Refprop					Aspen Plus				
	T (°C)	P (Bar)	h (kJ/kg)	x (−)	$\dot{m}$ (kg/s)	T (°C)	P (Bar)	h (kJ/kg)	x (−)	$\dot{m}$ (kg/s)
1	41.6	37	258.2	–	28.3	41.6	37	258.0	–	28.3
2	50.5	37	270.7	–	28.3	50.5	37	270.2	–	28.3
3	57.1	37	274.8	–	28.3	57.1	37	279.4	–	28.3
4	134.9	37	398.5	–	28.3	136.0	37	399.1	–	28.3
5	170.8	37	518.9	–	28.3	172	37	518.1	–	28.3
6	64.6	1.7	474.6	–	28.3	66.4	1.7	473.8	–	28.3
7	60.5	1.7	470.5	–	28.3	57.3	1.7	464.6	–	28.3
8	40.0	1.7	255.0	0.00	28.3	40	1.7	254.7	0.00	28.3
31	224.0	1.2	431.3	–	20.1	224.0	1.2	431.0	–	20.1
32	145.0	1.2	262.0	–	20.1	146.0	1.2	264.0	–	20.1
41	148.0	2.9	548.6	–	41.3	148.0	2.9	548.6	–	41.3
42	64.5	2.9	463.9	–	41.3	67.2	2.9	466.7	–	41.3
50	85.0	3.0	356.2	–	52.8	85.0	3.0	356.2	–	52.8
51	83.4	3.0	349.5	–	52.8	83.4	3.0	349.6	–	52.8
FWP power (kW)				92		92				
Turbine power (kW)				1253		1253				
Cycle net power (kW)				1161		1161				

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